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Considerations on Water Cooling of X-Band Accelerating Structures

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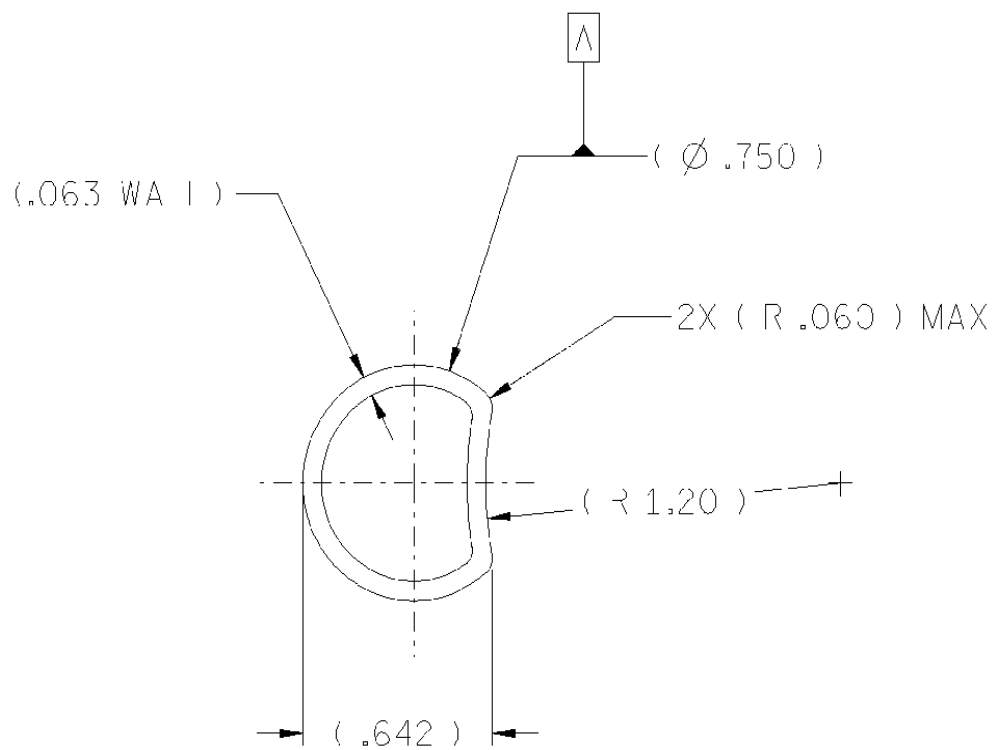
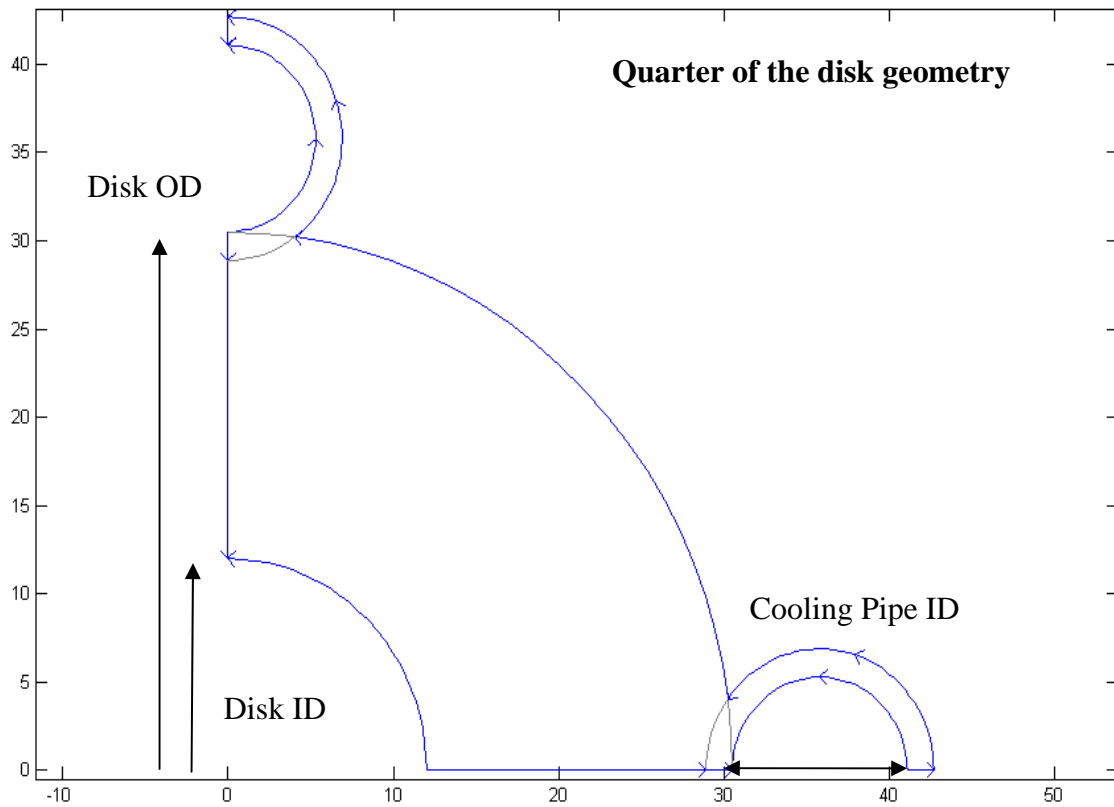
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The scope of this note is to present some calculations and considerations on the cooling of X-Band RF structures. Given the actual RF performance and the cooling configuration of the accelerating structures produced at Fermilab for the NLC, this note intends to address few questions in order to optimize the cooling system design of the accelerator. First the present geometry of the structure is studied, then the allowed temperature rise of the water and the cooling pipes diameter are modified in order to define their effect in the overall cooling. Finally an alternative configuration is proposed.

This thermal problem is solved using a 2D partial differential equation solver. The accelerator cavity is composed of several, in our case 53, copper disks brazed together and, given the cylindrical geometry, it is possible to solve the thermal equations in the cross-section and step by step extend the calculation through the sequence of disks. This model considers a simplified cross-section of the disk where the irises are ignored. The geometry of the model is shown in figure 1a and 1b compared to an FXC disk. A disk presents a typical outer diameter of 61 mm and an inner diameter of 24 mm with a thickness of 10.9 mm. Four tubes are used to cool the structure; their present dimension is 3/4" outer diameter. The tubes are formed, as shown in figure 2 in order to match the profile of the outer diameter of the disk, and allow for brazing. In the model configuration, in order to simplify the complexity of the calculation, the tube geometry and the presence of the brazing alloy are not considered. This simplification introduces some error in the calculations that is discussed later in this note.





Cooling Parameters

In order to generate the accelerating gradient of 65 MV/m, a structure needs a peak input power of 56MW. The effective length of the pulse is 400ns with a repetition rate of 120 Hz. Assuming that the lost power is around 63% of the input value, using the following equation 1:

$$P = P_{in} \frac{t}{0.8} f \frac{\eta}{L} \quad (1)$$

Where:

- P is the total power to be absorbed by the cooling water
- P_{in} is the input peak power
- t is the pulse length
- f is the repetition rate
- η is the percentage of the power adsorbed by the structure with beam off
- L is the length of the structure

it is possible to infer the heat load in the structure which is about 3.5 kW/m.

In order to operate correctly, the accelerating structure must be aligned to the beam with micron level precision. The present specifications for the main linac of the NLC call for a 50 μm vertical alignment structure to structure and a 100 μm within a girder (containing four structures) which is the fundamental unit of the main linac. In order to maintain this alignment it is necessary to minimize the thermal gradient due to the water cooling. This goal is achieved by choosing a counterflow configuration. In the ideal case of a cylinder with constant heat generation and counterflow outside cooling, the temperature distribution along the length of the cylinder remains constant. In order to apply this concept, two separate circuits are used where the two input pipes (and as a consequence the 2 output pipes) are positioned on the opposite side of the disks as shown in figure 3.

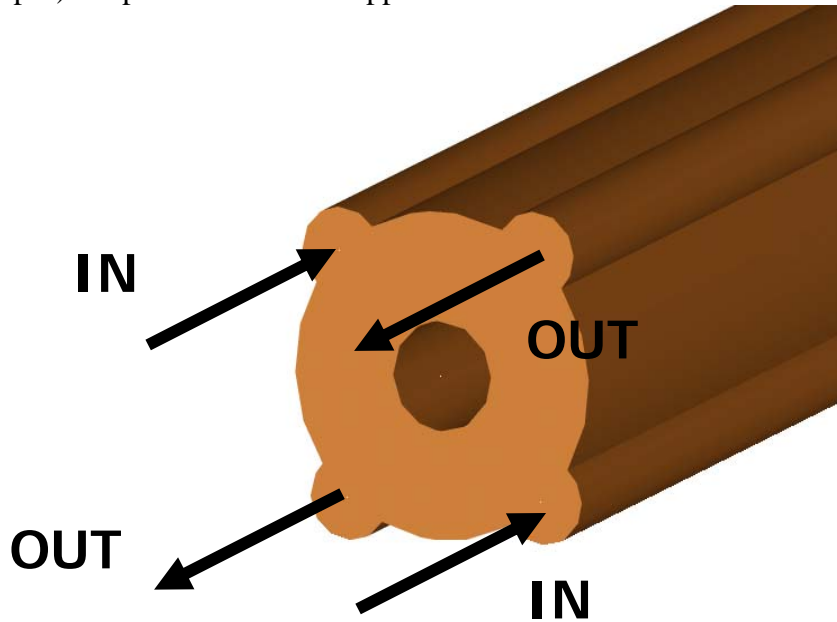


Figure 3 Counterflow configuration

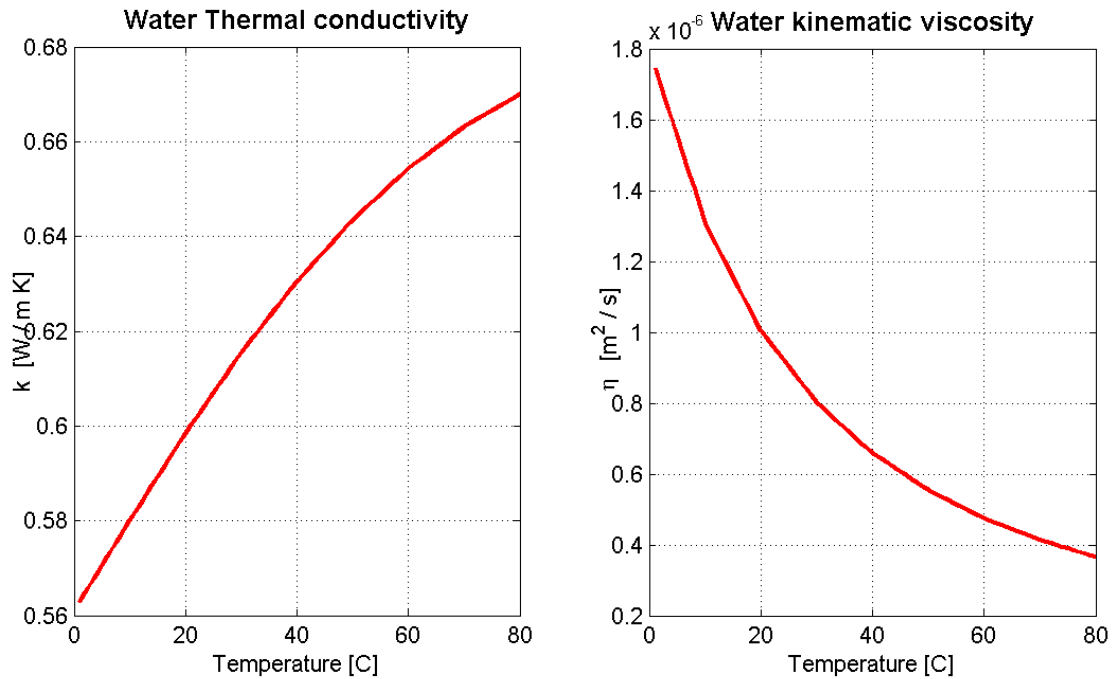
The baseline temperature rise allowed for the cooling water in the latest NLC design is 2 C. The average temperature of the RF wall of the structure (in our case the inner diameter of the disk) must be kept near 45 C. This value presented in the NLC baseline design results in the best electrical properties of copper [1].

The input temperature of the cooling water is defined by the whole accelerator cooling system. In the baseline design of the accelerator, cooling towers are used to compensate for the temperature rise of the water in the system. The optimal working point of these units depends on their geographical location. In this case, the worst scenario is the Californian site of the machine, where the lowest admissible temperature of the cooling water should be 82 F (~28 C).

A further consideration must be added on the vibration stability of the structures. The cooling flow has a substantial effect on the overall vibration level of the accelerating structures and the transmission of these vibrations to the adjacent elements in the accelerator. In addition to the optimization of the structure supports, reducing the cooling water flow rate is the next most effective way for reducing the vibration budget of this component.

Water Properties

In order to perform an accurate analysis over the range of temperatures that can be explored, the properties of water are calculated according to the local temperature. This methodology allows for a more precise calculation of the local heat transfer parameters along the accelerating structure.



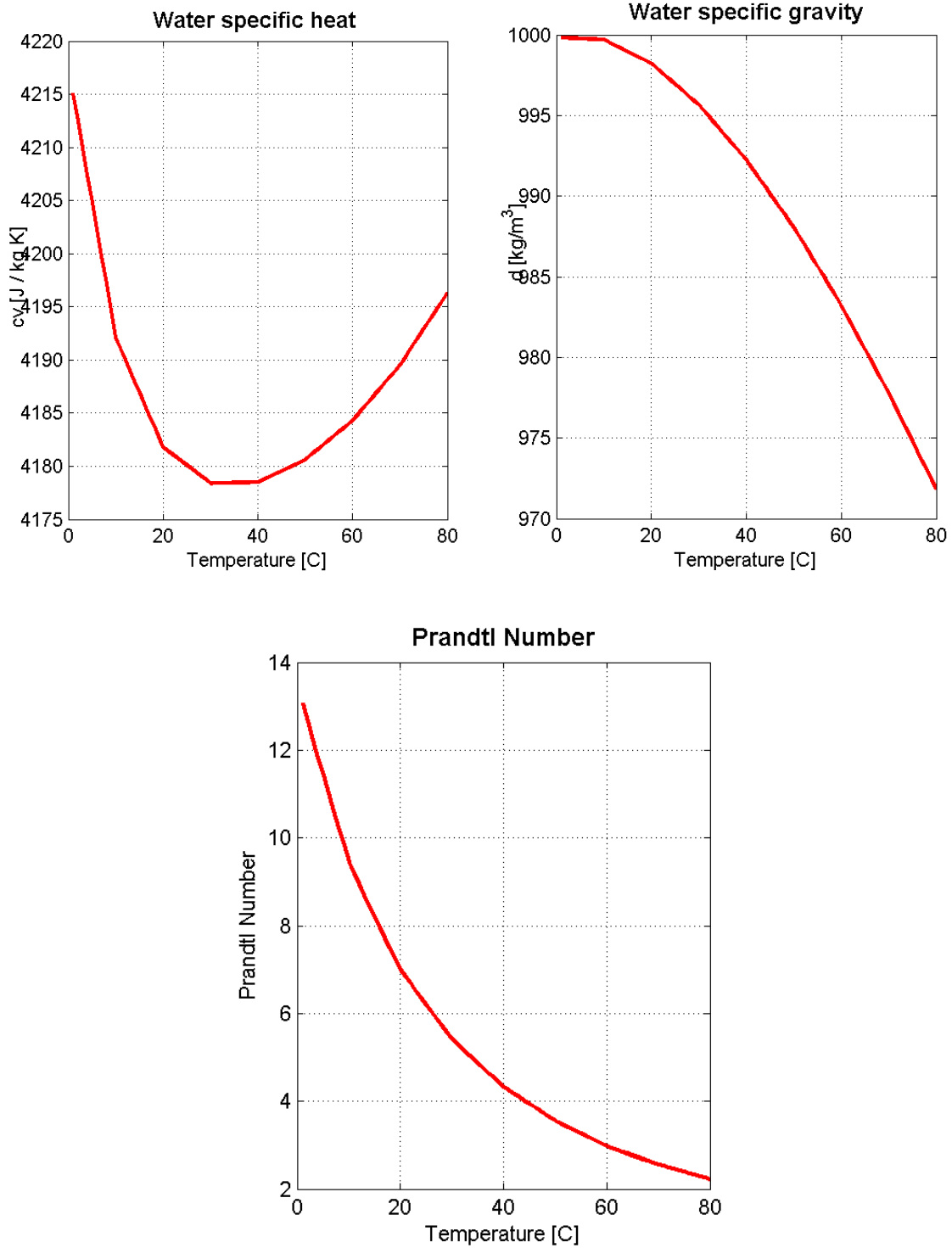


Figure 4 Water properties as a function of temperature

Calculation Procedure

The method used to calculate the heat transfer in the structure is well described in [2]. Starting from the pipe geometry, the heat load, and the desired temperature rise of the cooling water, it is possible to define the necessary water flow. As shown in equation 2

$$\dot{m} = \frac{P}{c_w \Delta T} \quad (2)$$

where:

- P is the power dissipated in the structure and adsorbed by the cooling water
- c_w is the water specific heat
- ΔT is the temperature differential in the water

all these parameters do not depend on the configuration of cooling or on the cross section geometry of the accelerating structure. Assuming a counterflow configuration, the thermal properties of the cooling water are calculated separately at the inlet and at the outlet pipes due to the temperature difference between the two to perform the calculations.

A MatLab finite element code is used to: generate the geometry and the mesh, and to solve the thermal conduction parabolic equations in 2 dimensions.

The mesh is generated only for one quarter of the structure cross section, due to its symmetry, and is obtained through a parametric geometry in order to allow for simulating different pipe diameters and thicknesses. All the thermal parameters are normalized to the disk thickness in order to reduce the 3 dimensional phenomenon to a 2 dimensional model.

At this point the boundary conditions must be defined.

The conditions on the two axes of symmetry imply a zero normal heat flow through them. All the external borders of the cross section are considered adiabatic. This is a conservative hypothesis since there will be a small heat exchange with the tunnel air temperature. The present design of the machine calls for a completely sealed tunnel during operations. Given the thermal capacity of the ground, it is possible to assume a constant temperature of the tunnel wall and a small natural convection between the air in the tunnel and the structure wall can be taken into account. As an alternative it is possible to extend this study in order to understand the implications of forced convection in the tunnel.

At the inner boundary of the disk a heat flux equal to the RF heat load must be taken into account.

The inlet and outlet water tubes inner diameters present a heat flux that depends on the local film coefficient and the difference in temperature between the water stream and the tube wall. For the first disk, the water temperature is fixed by the problem parameters. The solution of the thermal problem can now be calculated and the temperature distribution on the disk is defined.

In order to define the boundary conditions of the next disk it is necessary to calculate the heat load adsorbed by the water in the current step and calculate the temperature variation in the water stream. This is performed by analyzing the results of the temperature distribution on the disk, calculating the average temperature at the inner wall of the pipes,

evaluating the average heat exchange through the pipe surface and finding the water temperature rise. This process is done for all the disks that constitute a structure. The final result is the temperature distribution of the water and of the inner diameter of the disk along the structure.

Results

Fixing the input parameters as shown in table 1, one finds that the temperature at the inner diameter of the accelerating structure is around 41.6 C while the temperature distribution along the structure is almost constant as shown in figure 5.

The temperature distribution of the cooling water along the structure is shown in figure 6.

Power [W]	1750
Inlet Water Temperature [C]	30
Outlet Water Temperature [C]	32
Water Temperature rise [C]	2
Cooling tube ID [mm]	1.578
Water velocity [m/s]	0.5812
Total Flow [l/s]	0.227
Reynolds Number (30 C)	25149
Prandtl Number (30 C)	2.2197
Film coefficient [W/m² K] (30 C)	4453

Table 1 Input parameters

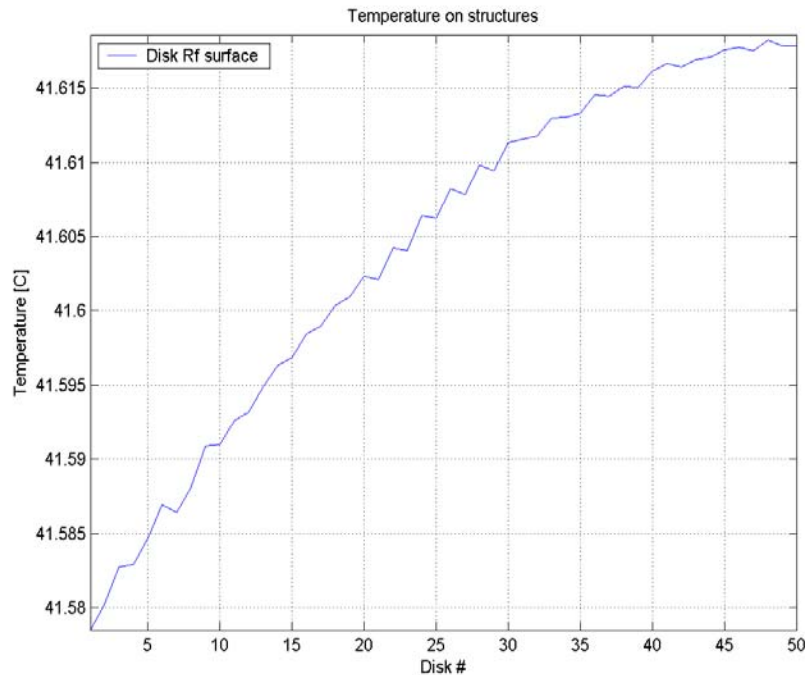


Figure 5 Temperature distribution along a structure with water differential of 2 C on counterflow

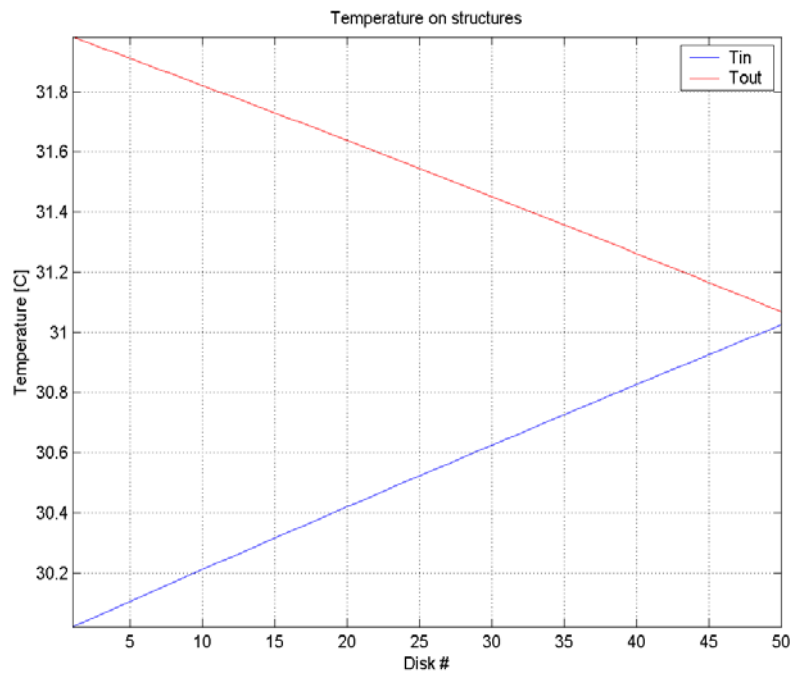


Figure 6 Temperature of cooling water along the structure

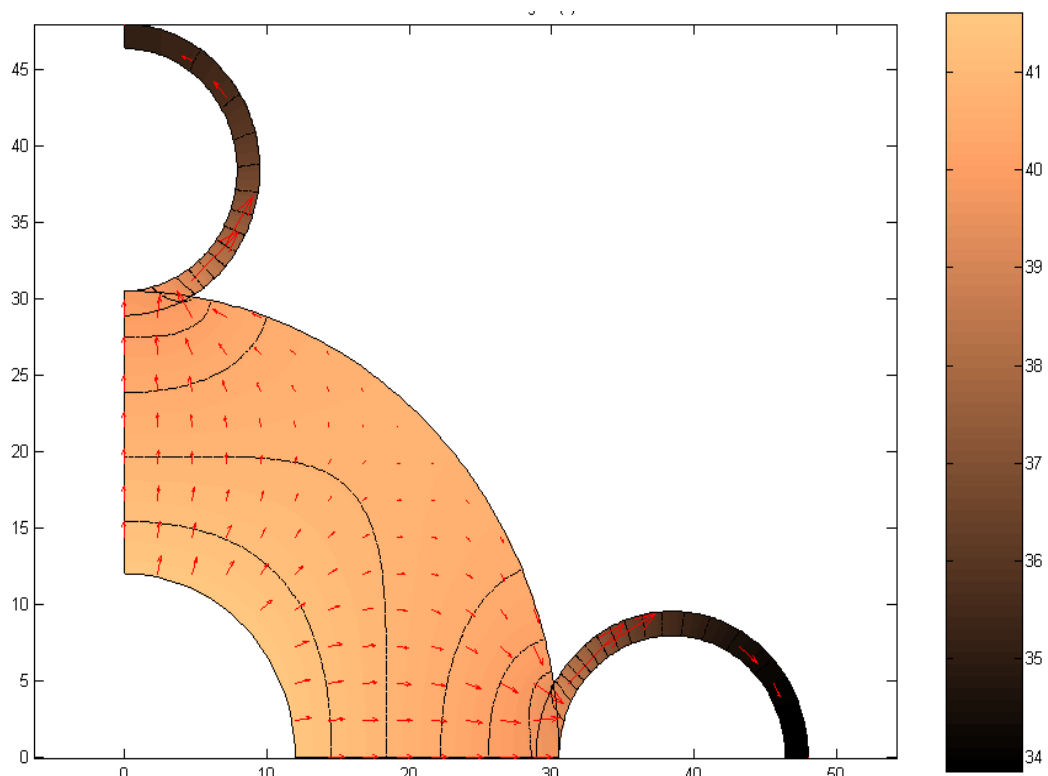


Figure 7 Temperature distribution in the disk cross section

The typical temperature distribution on a disk is shown in figure 7. From these results one can demonstrate that the temperature drop along the copper is very limited while all the temperature gradient is adsorbed in the water film.

Errors Due To Geometry Simplifications

The geometry of the model, as already mentioned, is simplified with respect to a real disk. While the iris area is not very important to be modeled precisely since the temperature gradient in that region, as shown in figure 7, is small; the area around the water cooling pipes present a high temperature gradient. In order to evaluate the effect of the simplification, a more precise geometry has been generated and the effect on a single disk was calculated. The result, as shown in figure 8, is an overall temperature drop across the disk due to better heat exchange between the disk outer diameter and the cooling pipe. Since the simplified model returns a conservative solution, it was chosen, at this stage, to avoid the more complicated mesh and use faster computing time.

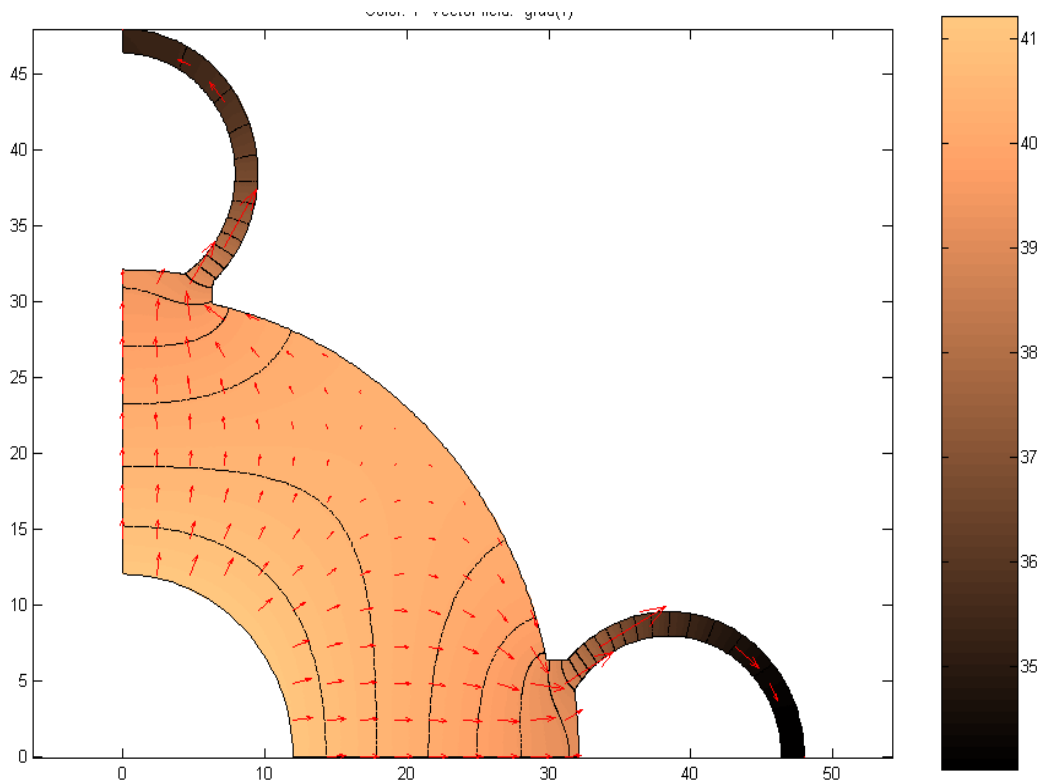


Figure 8 Temperature distribution in the disk cross section

Parameters Optimization

In order to optimize the cooling design, it was decided to evaluate the effect of single parameters such as the cooling pipe diameter and the temperature rise in the water.

These two parameters have a direct effect in both the heat exchange coefficient and the pressure drop along the pipes. In figure 9 is shown the effect of the cooling pipes

diameter. Considering pipes with half of the present diameter one can see a significant rise in the heat exchange coefficient, but the pressure loss along the pipe becomes too high. On the other end by raising the temperature differential in the water, as shown in figure 10, the flow is reduced with reduced pressure loss but also reduced heat exchange resulting in a lower initial temperature in the cooling water and higher costs for the all system.

An optimal solution would be to combine the two effects: reduce the dimensions of the pipes in order to obtain higher velocities and heat exchange coefficients, while raising the temperature differential to maintain the pressure loss below acceptable values.

A possible solution is shown in figure 11. In this case the temperature differential was raised to 3 C while the cooling pipes diameter was reduced from $\frac{3}{4}$ " OD to $\frac{1}{2}$ " OD. It is possible to see how the temperature distribution on the disk is not strongly changed while the reduced flow allows for smaller piping of the overall system and reduced vibration generation.

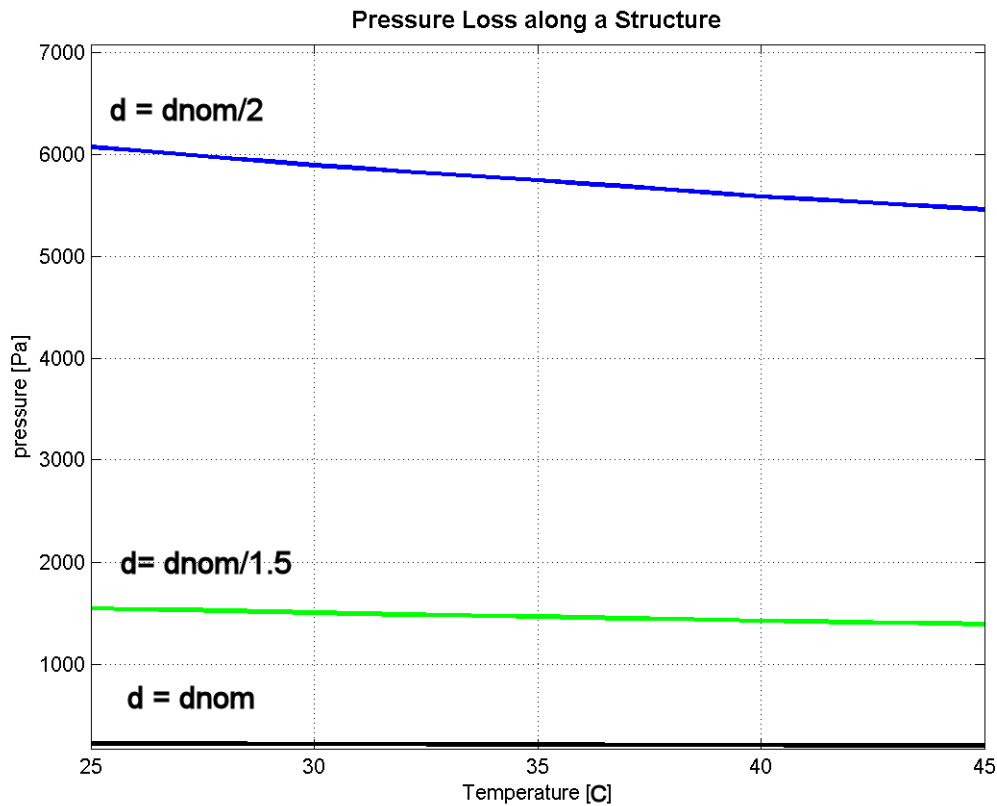


Figure 9a Pressure loss as a function of the cooling pipe diameter

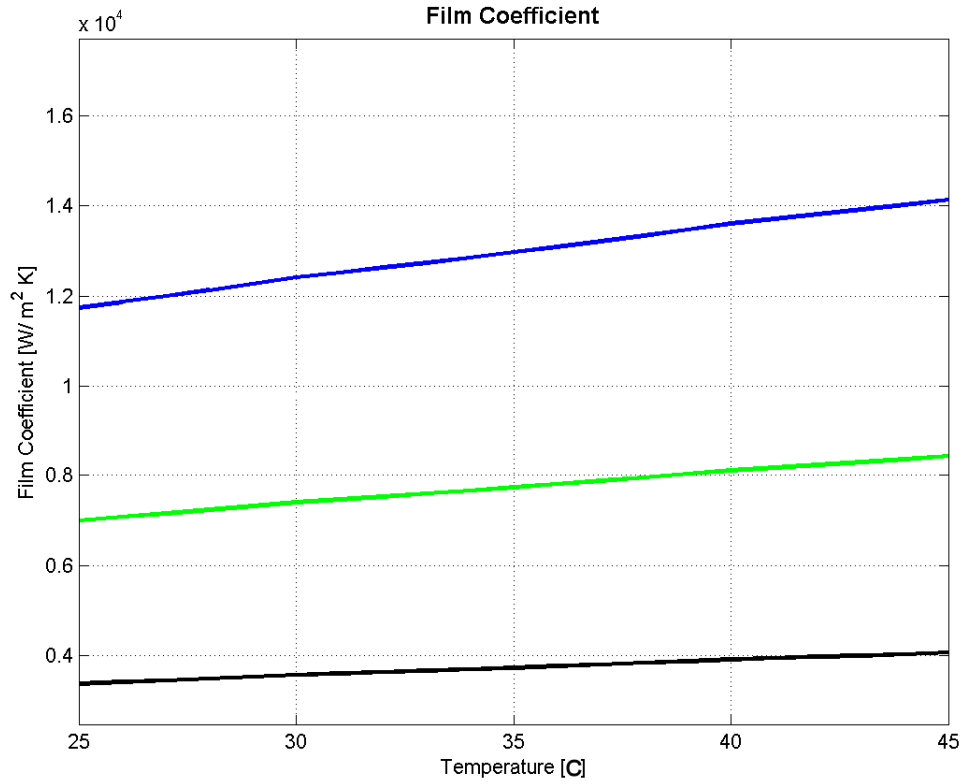


Figure 9b Heat exchange film coefficient as a function of the cooling pipe diameter

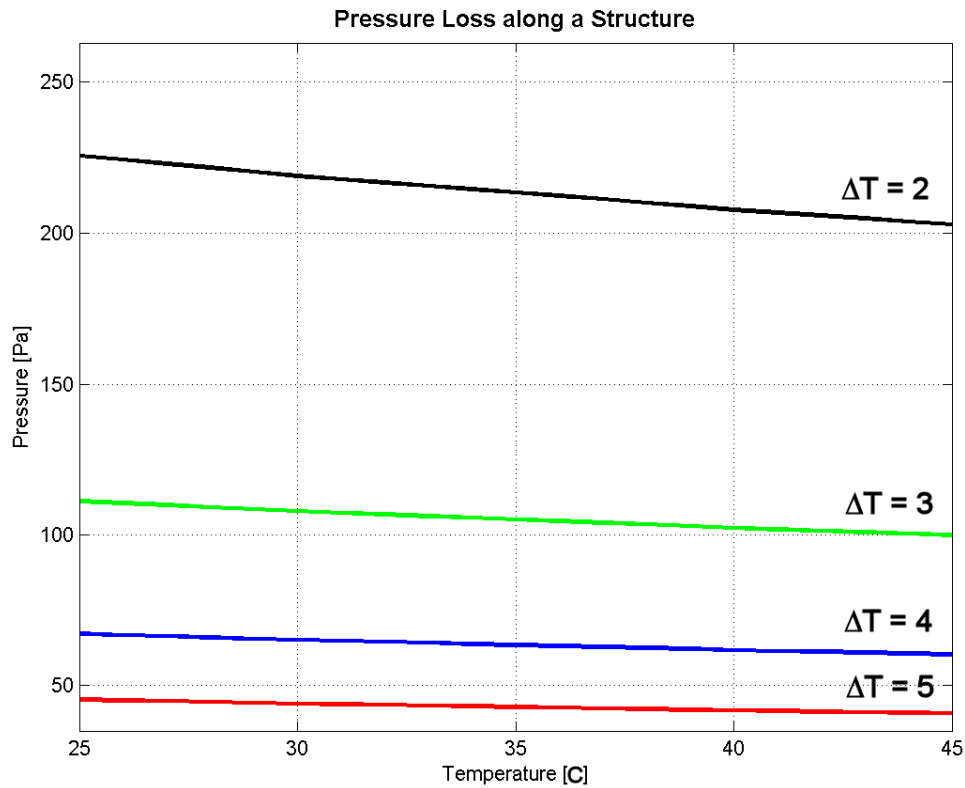


Figure 10a Pressure loss as a function of cooling water temperature rise

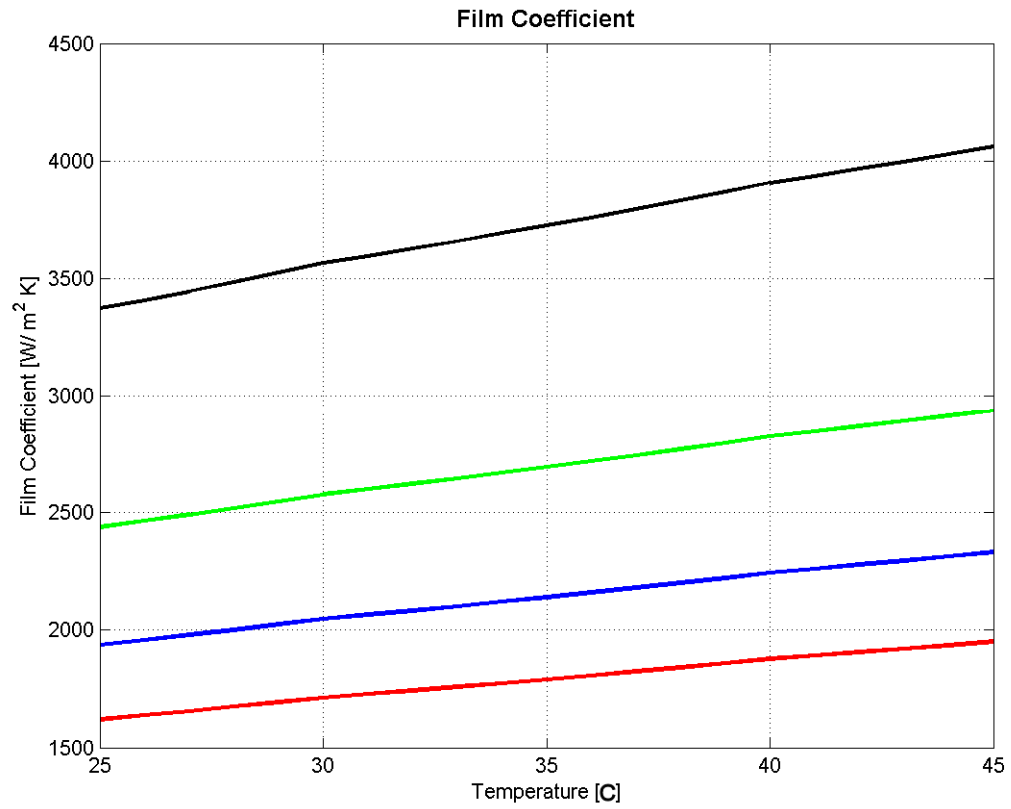


Figure 10b Heat exchange film coefficient as a function of cooling water temperature rise

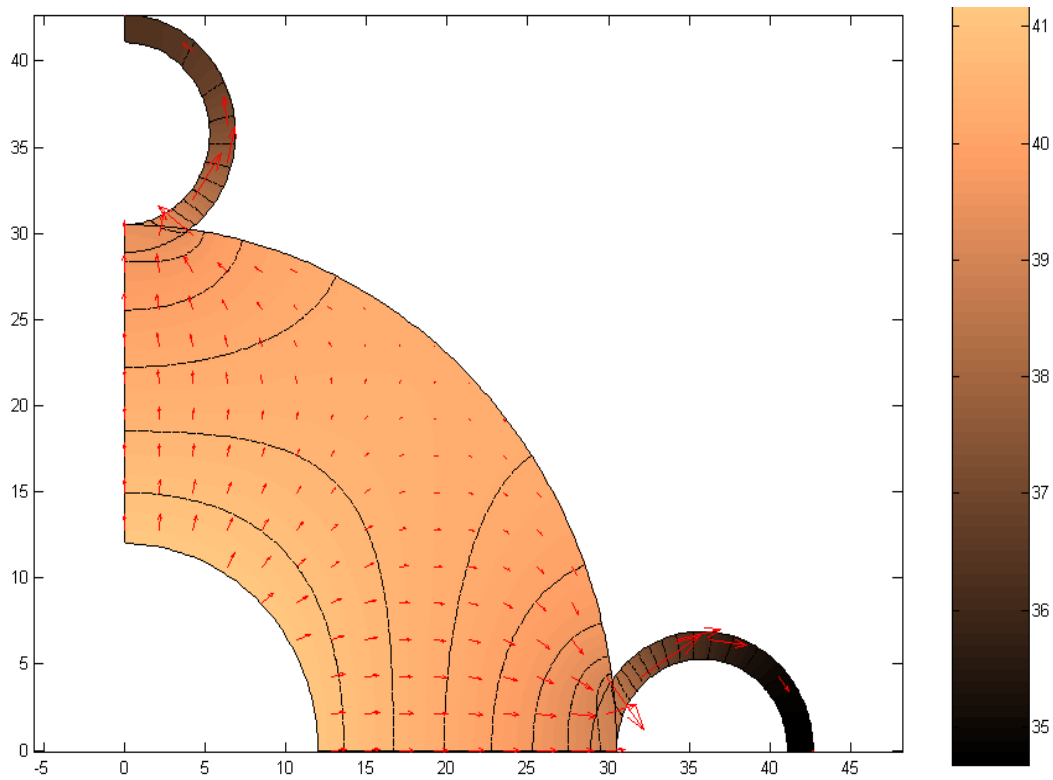


Figure 11a Temperature distribution in the disk cross section

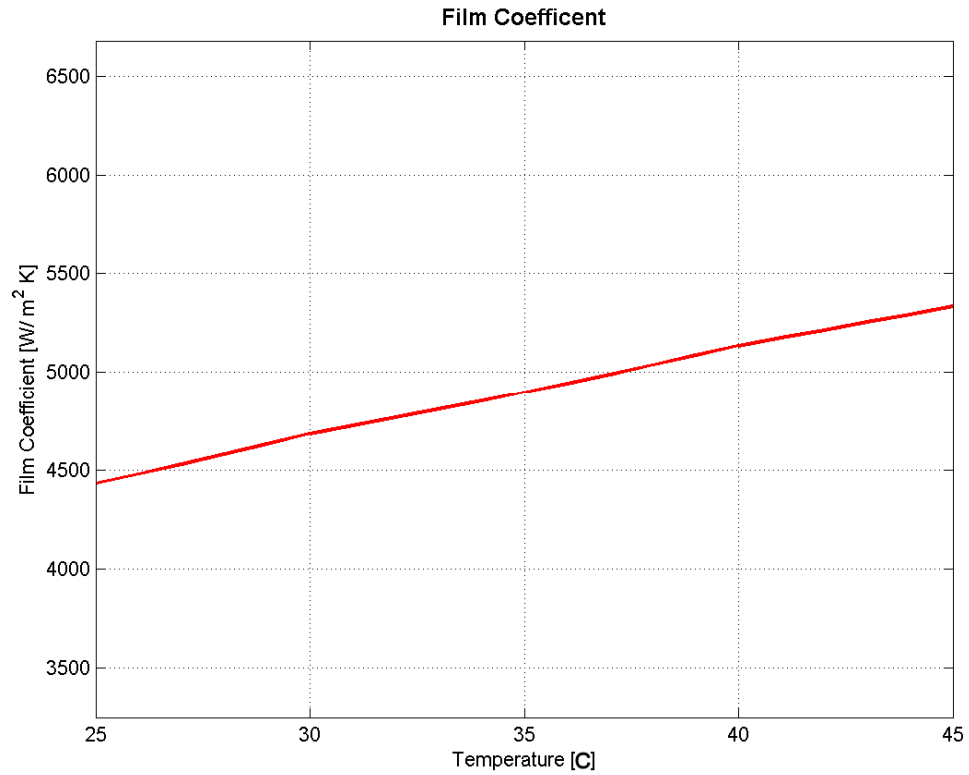


Figure 11b Heat exchange film coefficient as a function of temperature

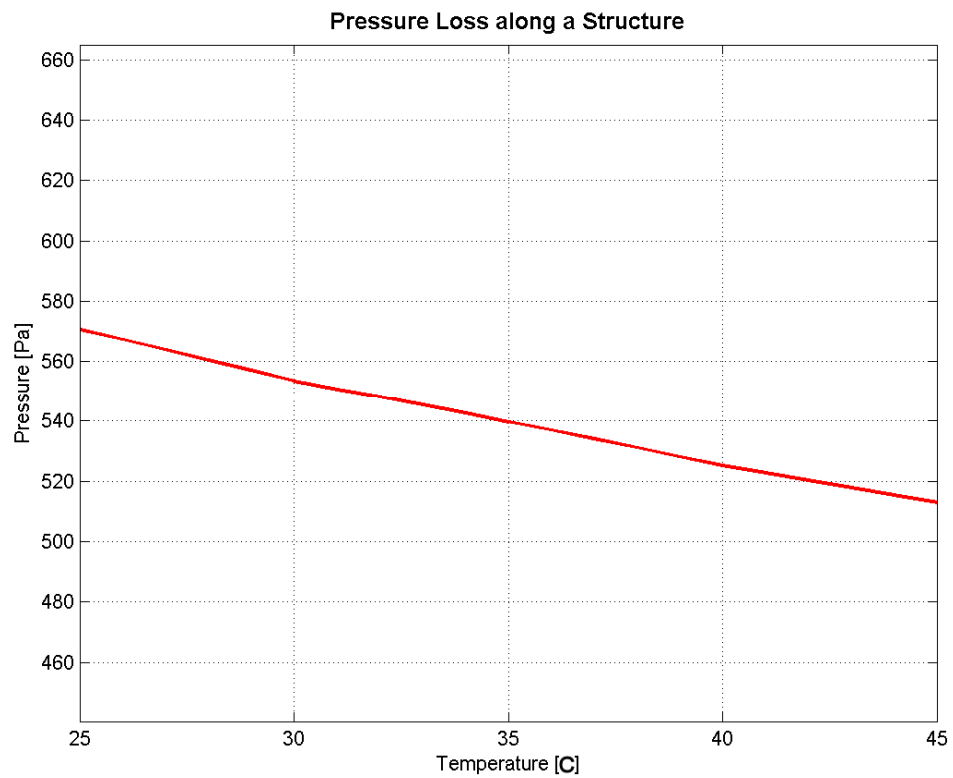


Figure 11c Pressure loss as a function of temperature

Conclusions

The problem of cooling X-Band RF structures has been presented. The present design of the cooling system has been analyzed and simulated using finite element code. Taking into account the effect of the single parameters affecting the heat exchange between the copper structure and the cooling water, a proposal for a more optimized solution is presented.

The following considerations can be outlined:

- The diameter of the cooling pipes should not be smaller than 1/2" in order to limit the pressure loss in the system to acceptable values
- The temperature rise in the cooling water should not be higher than 4 C in order to allow for optimal heat exchange coefficient
- Due to the high temperature gradient in the region, thicker wall pipes allow for better heat exchange
- Slots in the RF disks instead of forming the cooling water pipes would allow for better heat exchange due to higher area of contact.
- The water cooling parameters can be further optimized in order to reduce the pumping costs, pipes dimensions, and cooling towers requirements
- The design of the pipes can be optimized in order to reduce the pressure loss in the 90 elbows at the inlet and outlet of structures. Using slots in the disks instead of formed cooling water pipes allow to generate smooth transitions at the pipes inlet and outlet.
- A counterflow solution is the only one that can assure a stable temperature along the structure.
- Vibrations in the structure can be reduced by using higher temperature rise in the water, but one must be careful to maintain an optimal heat exchange coefficient in order to keep the water within a temperature raise suitable for heat exchange using a cooling tower.

Further investigations may result in additional parameters optimization and fixing the design.

8. References

- [1] NLC ZDR
- [2] G. B. Bowden, Low Flow Accelerator Cooling, NLC ME Note #20 09/26/2000